Applied Thermal Engineering 128 (2018) 107–120

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

A review of prechamber ignition systems as lean combustion technology for SI engines

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HIGHLIGHTS

- Concepts about homogeneous and stratified prechamber ignition systems are presented.
- Summary is provided for the main works concerning the prechamber ignition systems.
- Influence on combustion and emissions characteristics are discussed.
- Key advantages and challenges in prechamber ignition technology application are identified.

ARTICLE INFO

Article history:
Received 11 January 2017
Revised 21 August 2017
Accepted 24 August 2017
Available online 5 September 2017

Keywords:
Lean burn
Prechamber ignition systems
Torch-ignition
Emissions
Combustion

ABSTRACT

Use of lean or ultra-lean air-fuel ratios is an efficient and proven strategy to reduce fuel consumption and pollutant emissions. Previous works indicate that lean burn mixtures improves engine thermal efficiency by improving combustion quality, reducing heat transfer losses and increasing the possibility of apply higher compression ratios. However, lower fuel concentration in cylinder hinders mixture ignition, requiring greater energy to start combustion. To favor the ignition process, several high energy providers methods have been studied. Between them, prechamber ignition system presents potential reductions in emission levels and fuel consumption, operating with lean burn mixtures and expressive combustion stability. In this paper, a literature review has been made about prechamber ignition systems as lean combustion technology, focusing in the several investigations regarding combustion and emissions characteristics and presenting the key advantages and challenges in prechamber ignition technology application. From this review can be observed that the pre-chamber ignition system is an important way to increase thermal efficiency, reduce fuel consumption and emissions in spark ignition engines.

Contents

Abbreviations: ACIS, Advanced Corona Ignition System; APIR, Self-Ignition Triggered By Radical Injection; BMEP, Brake Mean Effective Pressure; BPI, Bowl-Prechamber-Ignition; BSFC, Brake Specific Fuel Consumption; BTDC, Before Top Dead Center; CA, Crank Angle; CFD, Computational Fluid Dynamics; CFR, Cooperative Fuel Research; CNG, Compressed Natural Gas; CO, Carbon Monoxide; COV, Coefficient of Variation; CVCC, Compound Vortex Controlled Combustion; DI, Direct Injection; EGR, Exhaust Gas Recirculation; GHG, Greenhouse Gases; H_2, hydrogen; HAJI, Hydrogen Assisted Jet Ignition; HC, Unburned Hydrocarbon; HCCI, Homogeneous Charge Compression Ignition; ICE, Internal Combustion Engines; IMEP, Indicated Mean Effective Pressure; $IMEP_n$, Net Indicated Mean Effective Pressure; IPCC, Intergovernmental Panel on Climate Change; JISCE, Jet Ignition Stratified Charge Engines; LAG, Ignition by Avalanche Activation; LPG, Liquefied Petroleum Gas; MAP, Manifold Absolute Pressures; MBT, Maximum Brake Torque; MC, Main Chamber; MFB, Mass Fraction Burned; NO_x, Nitrogen Oxide; PC, Prechamber; PCI, Prechamber Injection System; PFI, Port Fuel Injection; PJI, Plasma Jet Ignition; PSIE, Prechamber Spark Ignition Engines; RCM, Rapid Compression Machine; RF, Radio Frequency; SI, Spark Ignition; SKS, Stable Kernel of Combustion; TCCS, Texaco Controlled Combustion System; TGP, Turbulence Generating Pot; TJI, Turbulent Jet Ignition; VVT, Variable Valve Timing; VW, Volkswagen.

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http://dx.doi.org/10.1016/j.applthermaleng.2017.08.118
1359-4311/

1. Introduction

Climate and other atmospheric changes have prompted studies to minimize environmental impacts. According to IPCC (Intergovernmental Panel on Climate Change), the average global temperature could reach an increase of 6.4 °C by the century end [1]. To prevent impacts like this, changes in climate policies has been adopted, forcing engine manufacturers to reduce pollutant emissions levels. Transport sector is one of the largest greenhouse gases (GHG) emitters, and has motivated extensive researches and developments. With current policies, it is estimated that in 2030 the transport sector will be responsible for 75% of GHG emissions [2]. Therefore, studies have been developed to use renewable fuels and more efficient and "cleaner" engines. According to Kleeman et al. [3], vehicular use of biofuels, electrical energy and hybridization are part of the current market for low fuel consumption. Besides that, it is possible to operate with lean mixtures, up to a limit, without geometric changes in engines. It can be the path to reduce specific fuel consumption and exhaust emissions, allying benefits of better efficiencies through lower pumping losses in comparison to stoichiometric mixtures [4], mainly at medium loads. Despite the benefits of lean combustion, high cyclic variability resulted from lower burn velocity represents a major challenge which must be overcome, requiring a greater source of energy to ignite the mixture [5]. For this, lean burn ignition systems have been widely studied [6–12]. Among these systems, stand out the plasma igniters [8,9,13–16], laser-induced ignition [17,18], corona spark plug system [13,19] and prechamber ignition systems [19,20].

Several technologies form plasma to initiate combustion instead of a spark plug discharge, as Plasma Jet Ignition (PJI) [15,16,18], Railplug [12,21], Microwave-assisted Plasma Ignition [22–24] and Radio Frequency (RF) Plasma Ignition [25–27]. Compared to conventional spark plug, the plasma igniters have proved to be more rapid and allow the lean limit to be extended [14,27]. However, NO_x levels increased considerably [14,27], the igniters tend to suffer from erosion problems due to temperature rise [28] and the systems can cause interferences either in measurements and electronic control unit [24,27]. Laser-Induced Ignition provides faster combustion, higher engine power and lower specific fuel consumption if compared to the conventional ignition system [17,29]. On the other hand, higher levels of NO_x were observed, which is explained by higher flame temperature generated by the rapid combustion [18]. Advanced Corona Ignition System (ACIS) presents another method of high energy ignition for burning lean mixtures [14]. Comparing to the conventional ignition system, ACIS improves ignition efficiency, extends lean limit, improves cyclic variations and reduces HC emissions for low loads and idle conditions. However, it had no effect on other exhaust gases, and, at higher loads, the benefits of ignition system decreases, presenting no advantages if compared to conventional system.

Stratified charge ignition is an effective method to achieve the lean combustion, which had been proved by many researchers, with studies mainly focused on tumble flows [30], direct injection [31,32] and prechamber uses [19,33]. Though, prechamber ignition systems has demonstrated a reduction in combustion duration and an extension of the lean limit [33,34], and, unlike the other systems, NO_x emissions in this case are drastically reduced, due to lower peaks of temperature, reaching levels close to zero (<10 ppm) [35]. Whereas the emission reduction capacity of this technology and its effectiveness in burning lean mixtures, this paper aims to present a technical review about Prechamber Spark Ignition Engines (PSIE) concepts, summarizing the latest developments of this system, highlighting its influences on combustion and emissions, identifying advantages and challenges in prechamber ignition technology application.

2. Fundamentals of prechamber spark ignition

An effective concept to lean burn combustion can be a prechamber combustion system. Adams [36] investigated the use of prechambers to generate combustion turbulence and reduce burn time of lean mixtures, through the high energy flame jets from prechamber. According to Jamrozik [19], the use of prechamber systems usually consists in a very lean mixture, with λ factor above 2.0, entering the engine intake port and being aspirated to the cylinder, while a nearly stoichiometric mixture is added to the prechamber. When the prechamber mixture is ignited, large amounts of CO and HC are produced. With the pressure rise, the

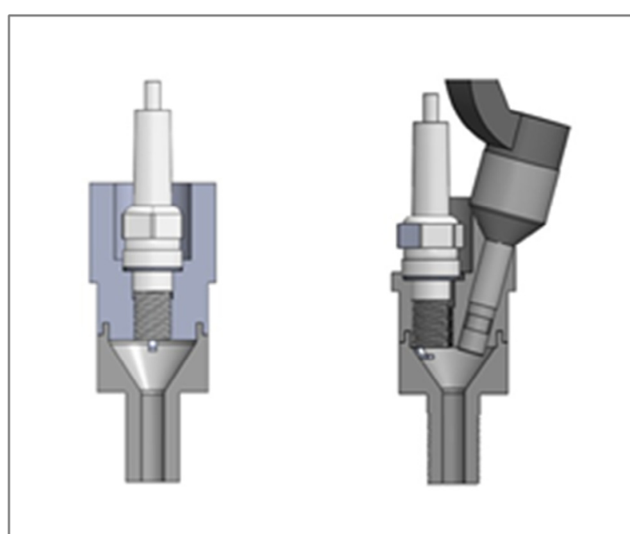

Fig. 1. Homogeneous charge prechamber system (left) and stratified charge prechamber system (right). Source: Author.

flame is forced to be pushed out through the connecting channel into the cylinder. Thereby, both flame and emission gases initiate the lean mixture combustion process. Particles of CO and HC are burned with the fast mixture ignition in main chamber and small amounts of NO_x are also produced. Despite the advantages in efficiency and emissions, lower energy supplied to the cylinder by the lean mixture results in indicated work reduction of up to 22.5%, in conditions of lambda between 1.4 to 2.0. Jamrozik et al. [37] emphasized that the indicated work can be increased by supercharging the engine and corrected by increasing the boost pressure.

According to the mixture into the prechambers, the system can be classified as homogeneous or stratified charge. In homogeneous charge prechamber systems, fuel injection occurs only in the main chamber, using either port fuel injection (PFI) or direct injection (DI), resulting in the same air-fuel ratio in both combustion chambers [38]. Then, air-fuel mixture enters in prechamber by interconnection orifices. Therefore, geometry of prechamber and connecting pipelines are critical for combustion development. Left side of Fig. 1 presents a homogeneous system, containing a prechamber and a spark plug.

Stratified charge prechamber systems are characterized by a fuel injection in the auxiliary chamber, usually DI, added to the main chamber injection, usually PFI. As the reduced volume of prechamber, a small quantity of fuel provides a rich region around the spark [39], being a solution to increase the stability and reliability of ignition and combustion process [40]. During compression stroke, the lean mixture formed in main chamber is forced into this prechamber, where the mixture is enriched with a small portion of fuel supplying [33]. Then, the purpose of this method is to benefit the combustion process creating regions with different air-fuel ratio inside the main combustion chamber, being richer at the vicinity of spark plug [41]. Right side of Fig. 1 presents a stratified system, with a fuel injector and a spark plug packaged in the prechamber.

Researches have been conducted to evaluate different prechamber ignition systems. Constructive variations are usually considered in order to improve systems efficiency, ranging size, geometry, air motion characteristics, placement and presence of valves or spark plugs [42]. Already in 1973, Wimmer and Lee [43] presented an alternative to ignite lean mixtures in spark ignition engines. They reported the use of very fuel-rich mixtures in prechambers to provide ignition of very lean mixtures in the main chambers. An air-fuel mixture with five times the stoichiometric amount of fuel was inducted though the prechamber intake valves at a rate of 2% of the main chamber mixture flow. Ignition process was started by a spark plug placed at prechamber.

In 1974, Davis et al. [44] have also realized studies of stratified charge engines, using the concept of Fig. 2a, designating the use of jet ignition principle as "Jet Ignition Stratified Charge Engines" (JISCE), and, in addition to experimental analysis, calculating the prechamber air-fuel ratio during compression and other combustion characteristics. A special cylinder head needed to be developed, incorporating two openings, one for a flush-mounted pressure transducer and other for a special thermocouple. Results pointed to adequately description of flow and combustion process.

In 1975, Gussak et al. [45] presented a study about Ignition by Avalanche Activation (LAG – initials from the first Russian letters of the expression which defines the process). The scheme of prechamber torch ignition and turbulent combustion is shown in Fig. 2b. A small prechamber near the combustion chamber is filled with a small amount of an auxiliary fuel-air mixture immediately before ignition, then, considering the high chemical activity of the products of incomplete combustion from a rich prechamber, occurs an fast, stable and complete combustion in main chamber. These products were emitted as a torch into the combustion chamber and burns completeness together with the whole amount of fuel. There was observed a period of initial flame formation in spark ignitions, whereas with prechambers there was a delay in fuel ignition. Gussak et al. [45] suggests that this system can be optimized using different prechamber and fuel injection characteristics. As results, were shown emission and combustion effects of different air-fuel mixture levels in prechamber.

Adams [36,46] developed two studies in order to relate torch chamber volume, orifice size, generation of turbulence and intensity of turbulence to burn rate. His first paper determines a theory for these relations independently of torch orientation, pointing to effects on exhaust emissions, power and fuel consumption. In his second paper, Adams [36] evaluated the effect of nozzle orientation on exhaust emissions, power and fuel economy, establishing the function between combustion interval and nozzle orientation in a torch ignited engine. Through the mechanism of Fig. 2c, the mixture which has been introduced to the cylinder in any of the conventional ways is compressed by the piston movement and forced though the orifice into the auxiliary chamber, creating a great deal of turbulence therein.

Lumsden et al. [47] presented an concept for combustion initiation using air-fuel mixtures as lean as $\lambda = 5$. Hydrogen Assisted Jet Ignition (HAJI) uses the fluid mechanics of a turbulent, chemically active jet, with the reliability of spark igniting rich hydrogen mixtures. In experimental analysis, they used a single cylinder engine, gasoline fueled with a supplementary system of hydrogen in the plane cylindrical prechamber, as shown in Fig. 2d. Hydrogen supplied between 1 and 3% of the main chamber fuel energy. For automotive context, was simulated the drive-cycle performance, achieving reduction in fuel consumption and exhaust emission levels.

Robinet et al. [28] presented a concept of firing system, called Self-ignition Triggered by Radical Injection (APIR – initials from the first French letters of the expression which defines the process), that allows to extend the engine operating range in terms of lean operating limit and thus of lean burn torque range. The device, shown in Fig. 2e, is mounted in place of the conventional spark plug, with volume of about 1% of the clearance volume and fuel supplied via a feed line with a rich fuel-air mixture. Interesting gains on fuel consumption in low load operating points were observed. In this scope, improvement of components was also responsible for achieving best results. In 1984, Latsch [48] presented a swirl-chamber spark plug which could accelerate the energy conversion process and achieve a further reduction of fuel consumption and exhaust emissions. As for this component [49], numerous patents have been developed over the years, ensuring properties to companies develop components [8,50–52] and technologies [35,53,54].

In terms of homogeneous charge concepts, Bowl-Prechamber-Ignition (BPI) [55] were presented as an alternative to high combustion temperatures and consequently high NO_x levels. The main feature is its dual injection strategy. First injection was done in the inlet stroke and leads to a homogeneous lean mixture, with air-fuel ratio between 1.4 and 1.7. In compression stroke a second direct injection with small amount of fuel is directed towards the piston bowl, as shown in Fig. 2f. The pressure difference between chambers and piston format favors a turbulent flow of the mixture into the prechamber. After the ignition of enriched mixture in the prechamber, flame jets penetrate into the main chamber and ignite the lean mixture. Mainly homogeneous lean mixture leads to low combustion temperatures and subsequently to low NO_x emissions.

Toulson et al. [56] evaluated an optical single cylinder engine operating with a Turbulent Jet Ignition (TJI) prechamber system. This generation of prechamber design, shown in Fig. 2g, simply

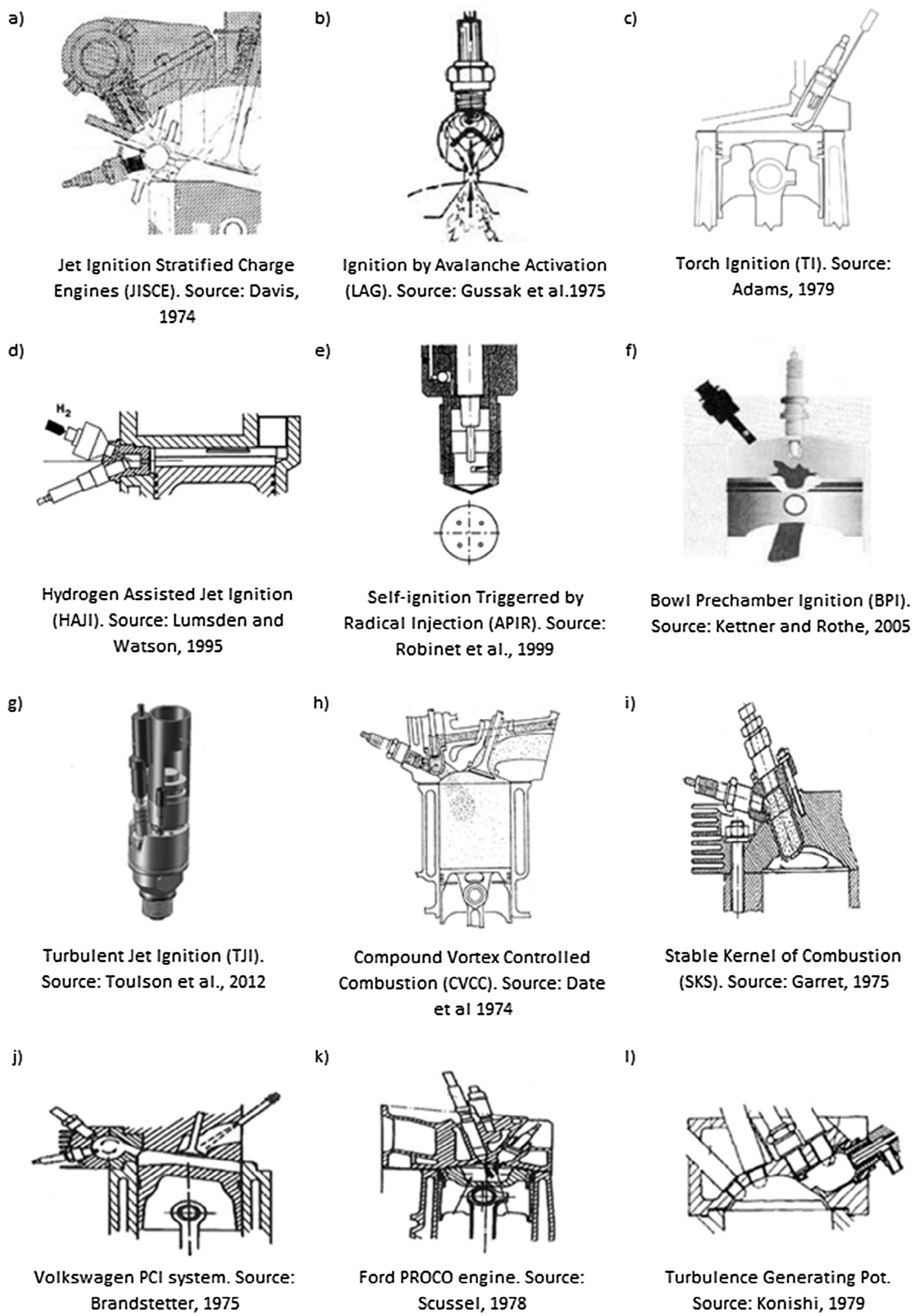

Fig. 2. Different configurations of prechamber ignition systems [7,28,36,44,45,47,56,73,74,76,80,81].

replaces the spark plug in a conventional spark ignition engine [57]. Prechamber combustion products were responsible to initiate combustion in the main chamber, enabling fast burn rates in multiple and widely distributed ignition sites. Ultra lean engine operation can exceed lambda 1.8 and short combustion durations were guaranteed according to the relatively small flame travel distances.

The use of computer simulation in engineering project development, such as internal combustion engines, is already established. Blair and Drouin [58] stands for time and test costs reduction, maximizing the time available to engine development. Especially in prechamber cases, Computational Fluid Dynamics (CFD) has allowed to understand how the behavior of a fluid flow is affected by its geometry and extend the analysis in a more complete way [59,60]. Two-dimensional simulations are a fast tool to investigate flow tendencies and patterns. However, the limitation to only two dimensions inhibits deeper analysis concerning the actual flow and combustion characteristics, especially with high levels of swirl and tumble motion. A solution to that is the use of 3D CFD simulations, as performed in many studies [61–67]. In those papers, authors investigated from differences of geometrical factors to fuel consumption and pollutant emissions. As an example, Thelen et al. [63] investigated the influence of ignition source positioning inside the prechamber on combustion speed and duration. Results showed that ignition source most far from connecting nozzle generated fastest jets and a more rapid combustion. Developments of hardware and computer models improved the quality of numerical analysis and made the tool more accessible to a greater number of researchers. But still, 3D CFD simulations are very time consuming if compared to other simulation methods, requiring boundary conditions which are only obtained experimentally, representing the major drawback to numerical simulations. Cruz et al. [68] argued that one-dimensional models can be used in addition to CFD simulations and experiments for engine analysis, achieving a more complete evaluation and improving the process comprehension. Thereby, Cruz et al. [68] described the development and application of a zero-dimensional mathematical model of compression and power strokes for a torch ignited engine. Several other recent studies point to computer simulation cases of lean burning combustion using prechamber [15,69–71].

With the need of reduce exhaust emissions and improve fuel economy, interest in commercial application of lean combustion engines also surged in mid of 1970s [72]. Fig. 2h presents CVCC (Compound Vortex Controlled Combustion), a stratified charge engine designed by Honda between 1968 and 1972 [73], with an auxiliary combustion chamber positioned adjacent the main chamber. By this controlled process, Honda has reduced NO_x, CO and HC emissions simultaneously. Addition of branch conduits into the torch passage contributed to reduction of hydrocarbon emissions, recirculating them back into the main combustion chamber for a secondary combustion event. With this, CVCC were able to meet primary emission targets of that time, with no detrimental effects to specific fuel consumption, and secondary emission targets with a large increase in specific fuel consumption. Fig. 2i presents Porsche stratified charge engine (SKS – Stable Kernel of Combustion), developed in 1975, that tried to create optimum conditions for each combustion phases [74]. In SKS, main mixture injection occurs in the inlet manifold whereas the prechamber injection is direct, with λ = 0.4 to 0.8 in prechamber and λ = 1.5 to 3.0 in main chamber, using the turbulence to increase the burn rate. Emissions of a standard 911 engine were reduced with specific fuel consumption remaining at acceptable levels, however, HC levels were still above those required by legislation. The Volkswagen (VW) Pre-Chamber Injection System (PCI), shown in Fig. 2j, is similar to the Porsche one, but uses swirl generated by the incoming charge from main chamber to increase the burn rate [75]. In VW concept, HC and CO exhaust emission legislation levels could not be achieved without after treatment systems and smoke also became a problem, though, a large reduction in NO_x was achievable. In 1978, Ford presented PROCO stratified charge engine with direct injection, which depends of swirl and injection timing to charge stratification, as shown in Fig. 2k [76]. In this concept was observed a slight reduction in specific fuel consumption and NO_x levels, relative to a standard carbureted engine, especially at part load. However, CO and HC were found to be higher. Texaco presented a similar design, however Texaco Controlled Combustion System (TCCS) varied the injection timing and rate to give the required power [77]. In TCCS, the heterogeneous nature of the combustion presented high smoke levels. Besides that, exhaust emissions targets were able to be met with the use of exhaust catalysts for CO and HC, and high levels of EGR for NO_x reduction. TCCS were similar with MAN FM engine [78], which injects the fuel onto the piston bowl surface and ignite by a spark plug. MAN concept presented lower peaks of pressure and a greatly reduction in CO and HC emissions. Fig. 2l presents concept developed by Toyota, the Turbulence Generating Pot (TGP) [79], with solenoid valves attached to sample the gases and intake oil valve to examine the effect of scavenging on NO_x emission. As other authors, Konishi et al. [80] concluded that the reduction of NO_x emission was due to the decrease of temperature gradient and lowered mean temperature of burned gases caused by these motions.

In addition of stationary points, transient conditions were also evaluated by some researchers. In 1974, Date et al. [73] have already studied CVCC behavior in transient cycles of federal standards. Results pointed to expressive reduction in CO, HC and NO_x emissions for three different engines if compared to the original ones. More recent studies of transient tests using prechamber concepts were also performed, as studied by Attard et al. [82], that obtained a successfully operation with jet ignition up to 3000 rpm and 10 bar of IMEP, near of the driving cycles operation loads, extrapolating it to transient conditions through computational simulations. Results pointed to more than 6% of fuel economy during NEDC and FTP-75 simulation when using homogeneous SI lean combustion and around of 13% when using jet ignition system.

In past applications the use of additional mechanical components, as three valves and two carburetors, increased engine production and maintenance costs, extinguishing prechambers uses in commercial engines. At that time, pollutant legislation requires small engine improvements [72]. With the electronic management prices reduction and increasingly stringent pollutant legislation, the outlook is favorable to the resurgence of prechamber ignition technology in commercial SI engines. Despite not being used commercially, the patented system of turbulent jet ignition prechamber combustion system for spark ignition engines [83] have been used since 2015 by Formula-1 racing teams, highlighting the ignition prechamber system as a current and promising alternative.

3. Influences of prechamber systems on combustion

Researchers [28,84–87] verified the necessity of provide more energy to start combustion and the low flame propagation speed as the main problems of working with lean mixtures. Use of PC ignition systems shows improvement in lean combustion ignition due to the greater amount of energy in main chamber at start of combustion. Increase in available energy for mixture ignition changes directly the lean limit, the spark timing, start of combustion, flame propagation speed and heat release rate. These essential combustion parameters will be discussed below.

3.1. Lean limit

The most advantages of PC ignition systems are related to the increased air-fuel mixture flammability limit, maintaining combustion quality and repeatability under stoichiometric mixtures conditions, when using lean mixtures on the main chamber. Moreover, systems with PC obtain expressive increase in fuel economy and low nitrogen emissions [88]. These conclusions were empha-

sized in a study with a Cooperative Fuel Research (CFR) engine performed by Wimmer and Lee [43] and in a divided chamber bomb studied by Yamaguchi et al. [89].

Flammability gains were studied by Toulson et al. [90], whose evaluated the lean limit of a Hydrogen Assisted Jet Ignition (HAJI) system in a CFR engine with gasoline mixture in Main Chamber (MC) and different gaseous fuels in PC. Results pointed that hydrogen (H_2) extended the MC lean limit to an air to fuel equivalence ratio (λ) equal to 2.5, while Liquefied Petroleum Gas (LPG), Compressed Natural Gas (CNG) and Carbon monoxide (CO) extended the lean limit to λ = 2.35, 2.25 and 2.15, respectively. This range of engine lean operating limit indicates that the improvement of fuel ignition levels on PC depends on several factors, including flame propagation speed and generation of chemically active combustion products, and not being only related with the amount of energy contained in the fuel [56]. Posteriorly, Toulson et al. [91] discussed about the use of LPG in both MC and PC, which would allow a single fuel system. Effects of have LPG or gasoline in main chamber and H_2 or LPG in prechamber were analyzed to determine the fuel influence over mixture lean limit, emissions levels and combustion characteristics. Fig. 3 presents lean limits for different fuel combinations, being LPG-H_2 combination the one with higher lean limit, followed by H_2-gasoline combination. LPG-LPG and LPG-gasoline were very similar cases and the difference between them was within the experimental error, not being considered.

Toulson et al. [90] reported that lean limits variations were higher at low manifold absolute pressures (MAP) than at high MAP and in all fuel combinations the lean limit was extended to slight higher than λ = 2.5. Lean limit extension differed over than 0.5 among considered fuel combination at low values of MAP. Generally the use of H_2 in PC extends the lean limit even more than when using LPG. However, there was no significant difference in lean limit extension whether LPG or gasoline was used as MC fuel.

Attard et al. [57] compared ignition and combustion characteristics in an optical single cylinder engine, natural gas fueled, in MC, using a stratified TJI system and providing information on the ignition process. Baseline lambda was changed from 1.3 to 1.8 when TJI system was implemented, achieving an acceptable combustion stability. Results indicated that the use of TJI with leaner mixtures produced a brighter and more intense blue flame than in the baseline engine. Authors attribute it to the increase of heat release rate that indicates a more stable combustion.

3.2. Spark timing

Combustion gases takes a period to expand and the angular or rotational speed of the engine can lengthen or shorten this time frame in which the burning and expansion should occur [92]. By this, the need to advance the time of sparking is associated with unburned fuel in the instant of spark fires. In general, PC ignition systems decreases ignition advancement which can be explained by an improvement in flame propagation speed. Since the three valve system with auxiliary carburetor [5,93,94], it was shown that combustion speed was significantly increased with PC use, where the ignition advancement for maximum power were 2 to 3 times lower than in an engine without PC.

Ryu [95] studied PC effects in spark timing of a homogeneous ignition system, correlating it with the intake jet direction for MBT condition. Conclusions pointed that the smallest spark timing angle occurred when the angle between the jet and the piston horizontal face was 90°. An APIR stratified prechamber ignition system was studied by Robinet [28], comparing the operation of a baseline SI engine and a system with APIR device. In Fig. 4 presents an optimization of spark timing for conventional burning case for MBT at 32° BTDC reducing to 10° BTDC with APIR system.

Roethlisberger [96] observed one of the most significant consequences of variation in spark timing delay, the expressive reduction in combustion temperature. Pischinger [97] either reported the reduction in combustion temperature, besides noting that this reduction has a negative impact over the Brake Mean Effective Pressure (BMEP) and Brake Specific Fuel Consumption (BSFC).

Fig. 5 shows the effects of ignition variations on Indicated Mean Effective Pressure (IMEP) and on combustion stability at 1500 rpm, with λ = 1.8, using a TJI system with constant air and fuel flow in a single cylinder engine [35]. Insensitivity of TJI combustion system

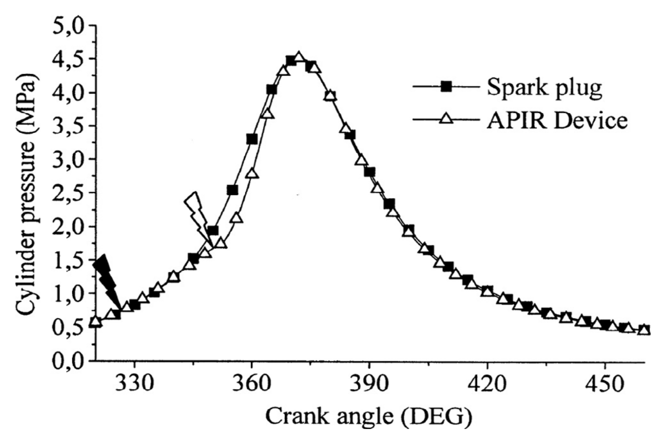

Fig. 4. Mean pressure versus each crank angle degree over 200 consecutive cycles [28].

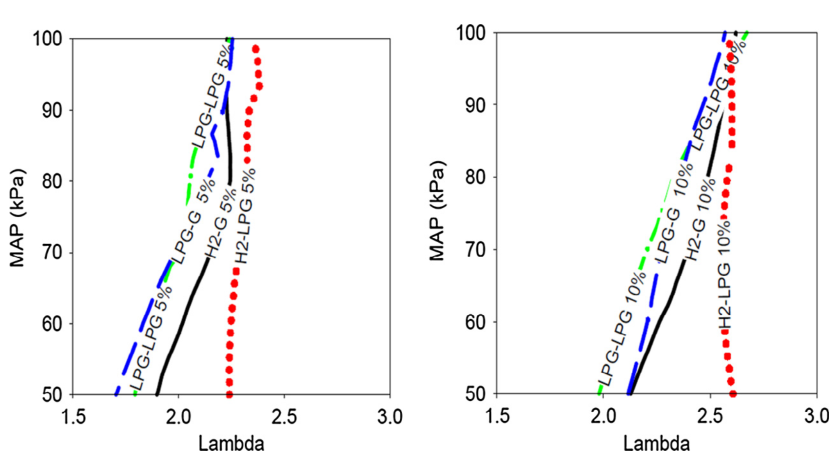

Fig. 3. Comparison at 5% (left) and 10% (right) COVIMEP of the PC-MC fuel combinations with varying λ and MAP (1200 rev/min, CR = 11 and MBT ST) [90].

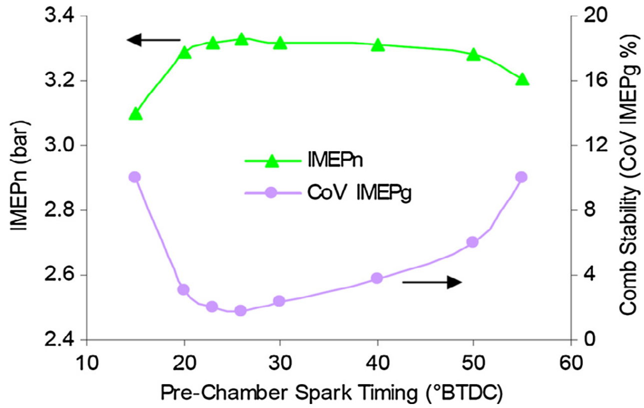

Fig. 5. IMEP and combustion stability for prechamber spark timing variation with TJI. 1500 rpm, constant air flow and fuel flow, λ = 1.8 [35].

to ignition variations when compared to conventional SI engine was observed. Also can be observed the low IMEP variation, about 1% over a range of 30° on spark timing, and combustion stability over a range of 40° on spark timing.

3.3. Start of combustion

Time duration expressed in crank angle degrees (CA) from the spark timing to the point when the pressure rise due to combustion is detected was defined by Sakai et al. [5] as ignition delay. Sakai et al. [5] and Ryu e Asanuma [6] observed that the ignition delay in PC ignition system was 11 °CA over all ranges of A/F, while the ignition delay in conventional engines was longer and increased gradually as the A/F becomes larger.

Start of combustion also can be analyzed by the Mass Fraction Burned (MFB). It represents the fraction of energy released from combustion by the fuel to the total energy at the end of combustion process, being determined from cylinder pressure analysis. Combustion rate affects thermal efficiency, cycle peaks of temperature and pressure, and exhaust emissions, usually being quantified in gasoline engines by the calculation of burn angles, which are the crank angles in which the MFB reaches a specified value [98,99].

Analysis developed by Toulson et al. [90] pointed to the influence of HAJI system in combustion start in a range of 0–2.5% MFB duration. As results, authors observed that combustion start is faster with H_2 than with other fuels, and that the flame initiation does not increase substantially with increasing A/F equivalence ratio. These effects were most likely due to the increased laminar flame speed and high levels of chemically active species present in the combustion of H_2 jet. Attard et al. [57] showed in his paper that the faster ignition in PC systems were due to the near constant mixture composition in PC added to the distributed ignition sites, provided by the jets and the high levels of chemically active species present in the combusting jets. For conventional engines without PC, more time is required to initiate and stabilize the flame kernel after the spark discharge due to the reduced kernel growth associated to the diluted mixture. Reduction in MFB duration for lean mixtures in a stratified PC system was also reported by Kettner et al. [7].

Gentz et al. [62] used the method presented by Shiga et al. [100] to compare 0–10% burn rate of a baseline SI engine with TJI system performance in a Rapid Compression Machine (RCM). It was observed that for SI system the propagation of burned fraction was slower than almost all TJI tested cases, indicating that TJI improved SI combustion start. According to Fig. 6, TJI system had a 0–10% burn duration longer than SI system when working with

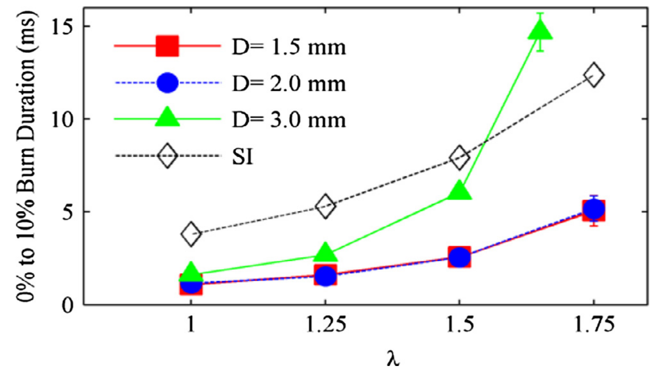

Fig. 6. Variation in 0–10% burn duration with λ for all TJI nozzle diameters an SI [62].

λ = 1.65 and with larger nozzle diameters. According Gentz et al. [62], mixture impoverishment and consequently jet speed reduction and combustion instability assigned the results of Fig. 6. Analyzing images of turbulent jet flame structure, authors observed that the jet development for smaller orifices resulted in increase of turbulence if compared with largest orifices.

3.4. Flame propagation speed

Literature shows that with the use of prechambers occurs an increase in turbulence, in available energy to start combustion, in radicals of active chemical species and in points of combustion initiation. According to Heywood [41], all these factors, which will be discussed below, contributes to increase the flame velocity. In order to verify if ignition PC system operating with lean mixtures was able to be equated or exceed the flame propagation speed for stoichiometric mixtures, Sakai et al. [5] conducted experiments on combustion duration for a single cylinder engine with and without PC. They found that the flame propagation speed becomes slower as mixture become leaner for both PC ignition system and the original one. Although they found that burn duration was greater for engine operation with PC for all A/F ranges. This result showed to be contradictory with most recent research. As showed in Fig. 7, Gentz et al. [62,101] found highly satisfactory results regarding burn duration for a RCM provided with PC ignition system.

Fig. 7 compares burn duration, using MFB 10–90% as evaluation parameter, for original system and three PC system applications, with different nozzle diameters. Gentz et al. [62,101] identified slight variations of burn duration between the different diameters

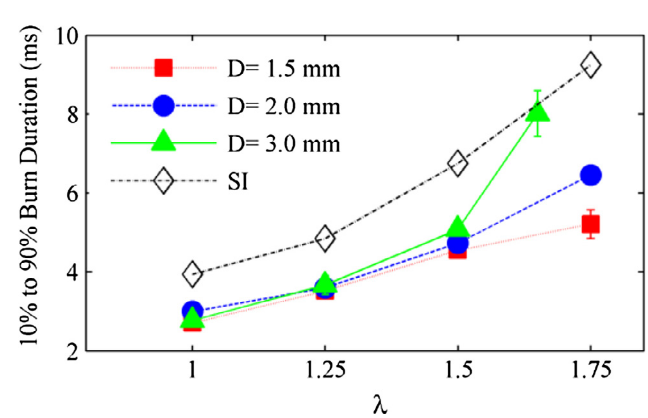

Fig. 7. Comparison of 10–90% burn duration between TJI system and the original one [62].

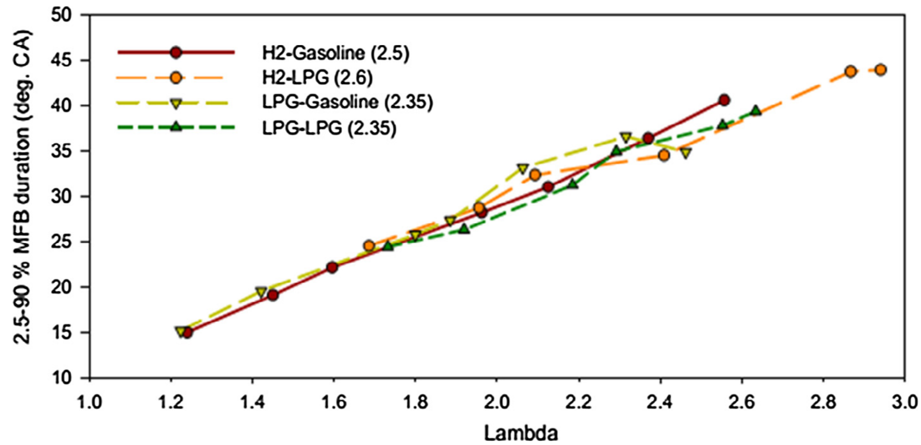

Fig. 8. Effect of PC-MC fuel combinations on the 2.5–90% MFB duration [91].

until λ = 1.5, at which point the burn duration increases as the diameter increases. This result means that, nozzle diameter had a reduced effect on flame propagation speed for conditions near stoichiometric. However, for mixtures leaner than λ = 1.5 the use of smaller nozzle diameters becomes advantageous due to turbulence increase, which generated more active radicals to initiate main chamber combustion. The use of PC ignition system proved be satisfactory since 10–90% burn duration was faster than the original one for all A/F ranges, indicating that the flame propagation speed was faster.

As Gentz et al. [62,101], Attard et al. [35] also agrees that the burn duration is faster using PC system than to the original ignition system. The 10–90% burn data indicates multiple flame fronts propagating through the MC. They justified that the shorter burn duration that occurs with jet ignition, was attributed to enhanced combustion provided by PC jets and the high levels of active radicals in it, being maintained in the propagating flame. Some authors identify the increase of flame propagation speed as a knock control agent [5,43,71,98].

Another factor explored about the flame propagation speed in PC engines was if it suffered any alteration due to the PC fuel. As shown in Fig. 8, Toulson et al. [91] compared burning duration, using 2.5–90% MFB as an evaluation parameter, for a TJI system operating with H_2 or LPG in prechamber and gasoline or LPG in main chamber. It was found that the burning duration was very similar for all fuel combinations, indicating that after the flame initiation process, the PC fuel has no longer effect. It suggests a similar jet penetration and a burning rate heavily dependent on the mixture contained in main chamber. In addition, the flame propagation and main energy release phase is very similar whether gasoline or LPG is used as the MC fuel. The authors also pointed out that the increased 2.5–90% MFB duration, for all fuel combinations, that occurs as the mixture becomes leaner was due to temperature drop which consequently lowered the flame speed.

3.5. Knock control

Higher compression ratios can lead to knock (spontaneous ignition of the end-gas due to the high temperatures) particularly at richer mixtures, where the charge is more reactive. This phenomenon is undesirable, since it reduces the power output and may also result in damage to the engine due to higher local heat transfer rates [71]. On the other hand, as knock results from self-ignition of highly compressed charge, long combustion time associated with lean mixtures are also minded for knock occurrence, unless the fuel contains sufficient anti-knock additives. In the meantime, these additives may contribute to increase exhaust emissions [46,59].

Although the use of prechambers makes possible the compression rate increase and the operation with lean burn, its use tends to minimize the occurrence of knock, since the end gas, that typically causes engine knock, can be burned before it ignites itself [34,61,62,82,102]. Hence, prechamber engines might exhibit lower octane requirement than the standard engines as shown by Wimmer and Lee [43] and Mehdiyev and Wolanski [103]. Some authors attribute the better knock control to the turbulence inside the main chamber [11,104], strongly related to the existence of PC system. Lower combustion temperatures from lean mixture ignition in main chamber, at full loads, also contributes to knock control [7,56].

3.6. Heat release rate

The heat release rate is defined as the rate at which the chemical energy of the fuel is released by the combustion process [105]. Many authors analyzed the heat release rate as a parameter on combustion process study in PC systems application. One of the first studies about this subject was made by Sakai et al. [5] as shown in Fig. 9.

It was observed from Fig. 9, a rapid increase in heat release rate at initial stage of combustion process and a decrease after a short stable period. Moreover, in the conventional engine the heat release rate shows a simple bell-shaped form. Sakai pointed to the first peak of heat release rate in PC system affected by A/F ratio inside the PC. With this, for rich mixtures in PC, the initial heat release rate showed a sharp peak and decreased steadily afterward. However, the peak value of heat release rate gradually decreases as PC mixture becomes leaner, being nearly of the conventional

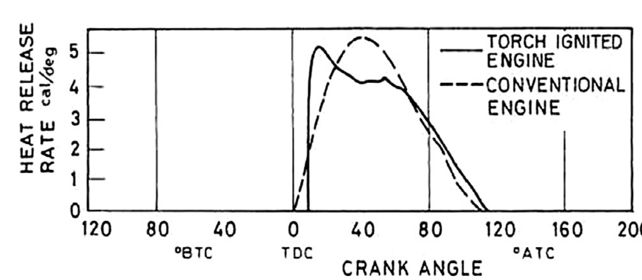

Fig. 9. Comparison of combustion characteristics at constant indicated specific fuel consumption. Adapted from [5].

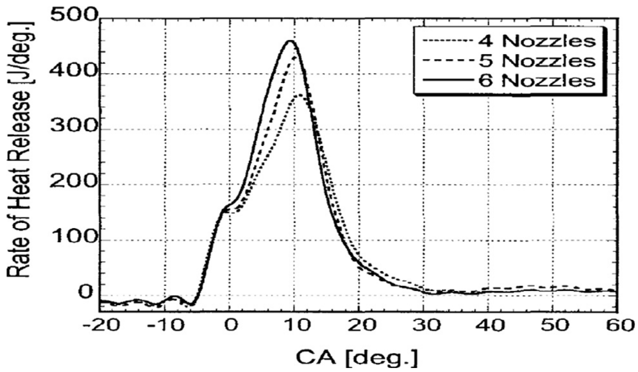

Fig. 10. Comparison of Rate of Heat Release in the main chamber [106].

engine and disappearing with the peculiar peak in the pressure diagram [5].

According Ryu et al. [95], the same behavior of heat release rate was observed for interconnection holes diameters between PC and MC. In smaller diameters, the heat release constant patch founded by Sakai was no longer detected and due to the high speed combustion, only the peak from first stage was identified. Ryu founded that the angle which PC flames enters in MC also affected the heat release rate. To vertical nozzle orientation, the crank angle of combustion finish was much earlier, and the rising gradient of heat release rate was steeper than for other torch nozzle directions, due to violent turbulence caused by torch jet impingement vertically onto the piston head [81,105]. As shown in Fig. 10, Kawabata et al. [106] noticed that a larger number of nozzles presents a shorter duration of heat release rate due to the promoted ignition and mixture combustion in MC.

3.7. PC geometry effects on combustion

As related by different authors, the improvement in combustion characteristics related to the use of PC ignition systems are strongly dependent of PC geometry and construction. Parameters as PC volume, nozzle number and diameter and the air-fuel ratio in MC and PC, modifies the behavior of the jet that enters in the main chamber, modifying combustion characteristics. As consequence, PC project is a hard task, requiring many times the use of computational tools.

Few methodologies are presented by authors to calculate PC geometry. First was showed by Adams [46] who said that the nozzle orifice size and auxiliary chamber, i.e. PC, were chosen in a way that the gases rushing out of the PC generates sufficient turbulence in MC to cause total burning of the mixture in an acceptable period of crank angle, not quenching the flame. He mentions that the most designs incorporate in PC, 8–15% of volume from the total clearance volume of the engine through a simplified zero-dimensional model.

Recent studies address to the use of computational tools to determine the influence of geometrical configuration as nozzle orifice diameter, number and orientation, as well as PC volume and internal shape [62,63,66,67]. But, as mentioned in Section 2, to achieve results with good accuracy, those tools are very time consuming and require high hardware's and software's cost. Another drawback is the boundary conditions that are acquired experimentally.

In order to reduce computational costs, the tendency is to use zero-dimensional and one-dimensional models to PC projects. Thereby, Cruz et al. [68] developed a concept of zero-dimensional computational model, aiming to analyze the combustion of a torch ignition engine. Computational model was capable to perform a parametric study of PC geometric design factor, being able to modify and test alternative parameters and to find the optimum value for PC volume and nozzle size for the engine used. Alvarez et al. [107] proposed a one-dimensional mathematical model to suppress the lack of methodologies for calculation of PC volume and nozzle diameter. Their model uses Adams [46] model as a basis but extend its use to different engines, conditions and fuels. From desired combustions parameters, the model performs numerical simulations varying parameters as PC volume, combustion duration, nozzle diameter and maximum specific energy provided by the PC (ϵ). A total of 17010 different arrangements were simulated. Authors mentioned that the maximum specific energy provided by the PC and maximum engine speed is highly affected by the orifice diameter. Therefore, considering a maximum speed of 6000 rpm and maximum ϵ presented in various arrangements was concluded that the optimum PC volume is about 7% of the MC's volume and the nozzle size should be of 6 mm.

Therefore, major limitations of burning lean mixtures can be mitigated by the use of ignition systems with a combustion PC. With this, as mentioned, the main chamber combustion is affected by increased lean limit, decreased spark timing for MBT, decreased time for combustion start, increased flame propagation speed, decreased knock and increased heat release rate.

4. Influences of prechamber systems on emissions

In last years, engine researchers has focused on improve engine efficiency to fit the constraints of regulatory standards. Major toxic components in engine exhaust gases are carbon monoxide (CO), hydrocarbons (HC) and nitrogen oxides (NO_x). Among these, NO_x are the most difficult to be reduced [20]. In this scope, ignition systems with combustion PC arise as a promising solution. Gas temperatures from combustion of lean air/fuel ratio in MC not support NO_x formation while maintaining low CO and HC emissions [19].

4.1. Emissions in stratified prechamber engines

In stratified PC engines, combustion starts in the fuel-rich PC with small formation of NO due to the lack of oxygen. As burning charge reaches the MC, the relatively cool mixture contained there promotes the rapid quenching of NO formation reactions. Furthermore, fast rate of energy release permits the use of retarded spark timing, contributing to decrease NO_x emissions [108].

Wimmer and Lee [43] studied the application of combustion PC combustion in a research engine and found more than 50% of reduction in NO_x emissions when compared with the original ignition system, both with λ = 1.2. NO_x emission values less than 1 gram per indicated horsepower-hour (g/ihp.h) were found for λ = 1.8. CO emissions were reduced by 82% when compared with the original system, while HC emissions were higher by approximately 83%. This increase was attributed to the quenching of oxidation reactions in combustion chamber wall region. Promising results were also found by Davis et al. [44] for the JISCE system with a 25% leaner mixture, resulting in a reduction of over than 80% in NO_x specific emissions keeping CO specific emission levels.

PC geometry influences on emissions was extensively studied by several authors. Sakai et al. [5] investigated the influence of nozzle's diameter of a PC with 10% volume of the MC in a single cylinder engine varying the nozzle diameter from 4 to 13 mm. It was found that, CO and NO_x emissions have no significant changes with variations of nozzle diameters. However, at part-load conditions, HC emissions increased considerably with the diameter increasing. They explain this increase by the fact that with greater diameter, more burned gases from the PC rich mixture are allowed in the MC.

Wall and Heywood [42] experimented two PC designs, one with 8.7% of MC volume and other with 12%. Results showed that NO_x emissions decrease as PC volume increase, while HC and CO emissions were little affected. Dimick et al. [109], varying the PC volume from 4% to 12%, found that the 8% volume had the best compromise between HC and NO_x emissions.

Varying nozzle diameter from 8 to 12 mm, Wall and Heywood [42] found that the indicated specific NO_x, HC and CO emissions tends to increase with orifice size. Lower emission values from smaller orifice diameters were result from more complete mixing between quench zone and bulk gases during the combustion process. Similar results were found by Dimick et al. [109].

Shah et al. [110] investigated the influence of nozzle diameter and PC volume on the NO_x specific emissions in a large SI engine fueled with natural gas. In contrast with results of Wall and Heywood [37] and Dimick et al. [109], they found that NO_x levels increased with the reduction of nozzle diameter.

Other relevant influences of PC geometric parameters on emissions was researched by Adams [36]. The configuration proposed can be seen in Fig. 11. To better understanding, this figure shows the cylinder head as viewed from within the combustion chamber, which was divided into twelve equal segments. These segments can then be linked to the face of a clock, where a line drawn from the center of the torch chamber through the geometric center of the cylinder head establishes the 12 o'clock position. These segments are then numbered clockwise from this position, so that the torch chamber can be manually rotated to the desired position by loosening a locking nut and discharge the burning jet in any direction over 360 degrees. Because of the proximity of the wall in the 5, 6 and 7 o'clock positions, only five directions were chosen: 8:30, 10 o'clock, 11:30, 1:30 and 3 o'clock position. After test these various nozzle positions, it was observed a change in HC and NO_x emissions of approximately 300% depending of the nozzle orientation. Minimum emission values were found for the 8:30 PC nozzle position.

Yagi et al. [111] reported the results of HC and NO_x emissions for the PC system, adapted from the CVCC engine. A reduction of 20% was observed in HC emissions for the engine with PC system in relation to the original CVCC engine, both equipped with PC. Similar results were found for NO_x emissions. These reductions are explained due to the exhaust gas recirculation provided by PC system, resulting in reduction of peak pressure and temperature in the main chamber.

Effects on emissions in engines operating with HAJI system have been extensively studied. Lumsden and Watson [47] found an reduction of approximately 98% in NO_x levels and about 60% in CO levels when compared with the original engine. Hamori [112] studied the application of HAJI system in a supercharged engine and observed, for the whole operating range, a reduction of 90% in CO emissions, an increase of 3.5 times in HC emissions when NO_x levels were near zero. Meanwhile, Toulson et al. [87] studied the effects of hot and cool EGR on specific NO_x emission for the HAJI system, concluding that in both applications, NO_x emission values were lower than when operating without EGR.

Other relevant research on HAJI system was made in order to study the effects of fuel on emissions [90,91,113]. These studies showed that NO_x emissions were not strongly influenced by the fuel contained in the PC. Specific HC emissions have varied up to 70% depending of the fuel used in PC, yielding lower values with the use of hydrogen. CO specific emissions were also changed according to the fuel in PC. An important observation was made on the use of CO as PC fuel, as there is no significant increase in specific CO emissions, it indicates that a complete burning was present in the PC.

Kettner et al. [7] compared an engine with and without BPI system and analyzed the EGR influences on indicated specific HC and NO emissions. Emissions of HC increased approximately in two and three times for the BPI engine, with and without EGR, respectively. However, NO emissions for the BPI system were approximately 78% and 74% lower than for the original one, with or without EGR, respectively.

Significant emission reductions were found by Attard et al. [83] in a single cylinder engine working with TJI system. Fig. 12 shows the comparison for NO_x emissions. Potential reductions were achieved with TJI system after λ = 1.1, reaching levels close to zero (<10 ppm) until the combustion becomes unstable at λ = 1.8. Authors explained that this emissions reduction was due to lower peaks of temperature when TJI system was used, and pointed out that, through this application, aftertreatment systems for NO_x reduction would be unnecessary.

Fig. 13 shows a comparison of CO emission made by Attard et al. [83]. It is observed that for both conventional ignition and TJI system, the percentage of CO emissions increases as combustion becomes unstable, which occurs for λ = 1.4 with the conventional system, and for λ = 1.8 with the system TJI.

Comparison of HC emissions made in the same study appears in Fig. 14. Minimum values for the two ignition systems are in the lambda range of 1.1–1.2, increasing to both systems for richer or leaner mixtures. Attard et al. [83] exposed that for richer mixtures it increase in HC emissions was due to incomplete combustion, while for leaner mixtures the increase was due to lower combustion temperatures, leading to reduction in wall temperature of

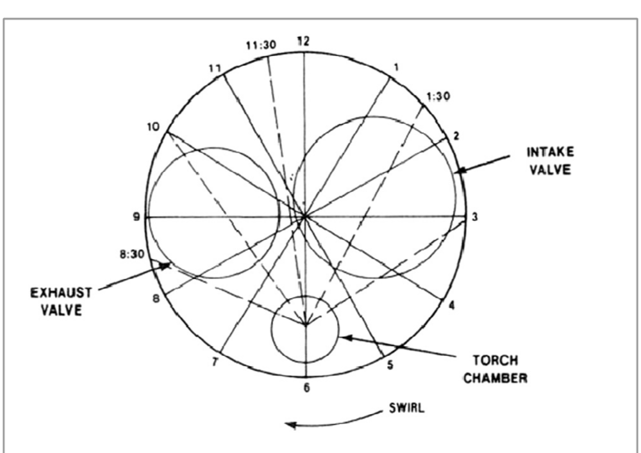

Fig. 11. Rotational positions of nozzle orifice as viewed from within the combustion chamber [36].

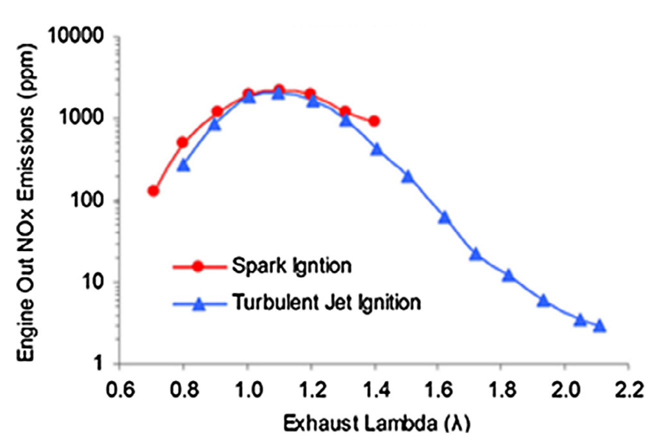

Fig. 12. Comparison of NO_x emissions at constant speed/ load worldwide mapping condition of 1500 rev/min, 3.3 bar IMEPn (~2.62 bar BMEP) [83].

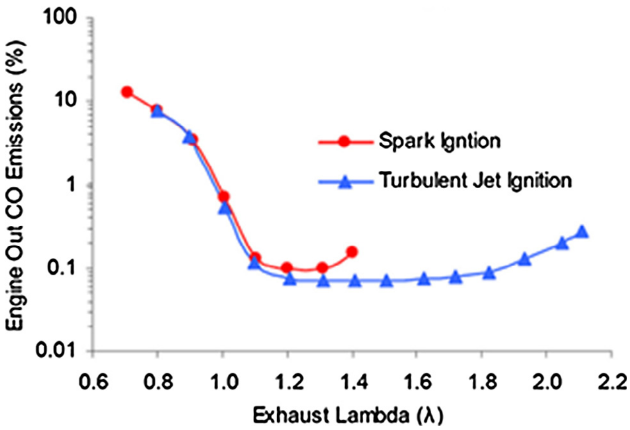

Fig. 13. Comparison of CO emissions at constant speed/load worldwide mapping condition of 1500 rev/min, 3.3 bar IMEPn (~2.62 bar BMEP) [83].

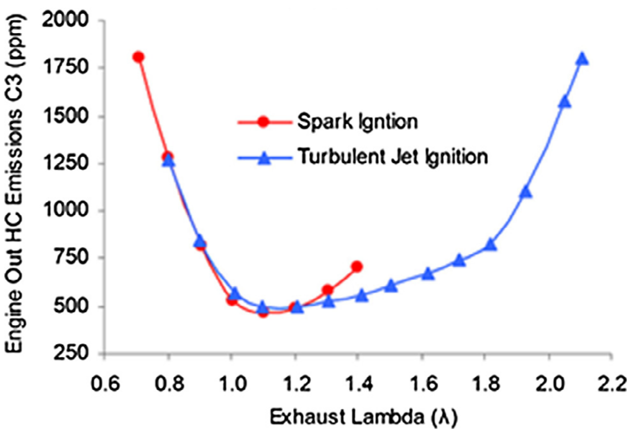

Fig. 14. Comparison of HC emissions at constant speed/load worldwide mapping condition of 1500 rev/min, 3.3 bar IMEPn (~2.62 bar BMEP) [83].

the combustion chamber and to consequent reduction in the burning of hydrocarbons during the expansion and exhaust strokes. It is also observed that the minimum amount of HC emissions was greater for the engine operating with TJI system. This was explained by the increased amount of crevices and bigger surface area due to the presence of the PC.

Rodrigues Filho [114], in studies conducted at the Federal University of Minas Gerais (UFMG), presented the results of emissions from a prototype with PC ignition system operating with different levels of stratification and several engine loads and speeds. Author found an average reduction over than 71% in CO specific emissions for operation with λ = 1.3, and a maximum reduction of 87% at 2500 rpm and 47% of maximum torque. The specific NO_x emissions had an average reduction over than 49% at λ = 1.3, whereas the HC specific emissions had an average increase of 33%. The author attributes this increase in hydrocarbon emissions to the formation of a fluid fuel film on the combustion PC.

It was observed that in most studies with stratified PC engines, the emissions results found were promising, especially when used in conjunction with other technologies. The main reductions were found in the nitrogen oxide concentrations, which reached almost zero levels. Three factors that have most effect on formation of NO_x emissions were the peak combustion temperature, the residence time at high temperatures, and the oxygen availability. CO emissions were also reduced, but HC emissions were higher in most of studies. Lawrence and Watson [115] found that active radicals help to decrease the level of quench and piston crevice sourced HC, but further studies are needed in order to identify the cause of this increase and to treat or control such emissions.

4.2. Emissions in homogeneous prechamber engines

In homogeneous prechamber engines, the jet flame emerges from PC and creates turbulence in MC, accelerating the burning speed. The bigger turbulence seems to increase the cooling loss from the chamber wall, decreasing the gas temperature. The earliest burned gases has higher temperature than the last gases to be burned. The TGP system tends to decrease this temperature gradient, and together with the cooling loss, tends to further reduce the NO_x formation [80,116].

Konishi et al. [80] analyzed experimentally and theoretically the NO_x formation process for a Toyota engine with TGP system. They analyzed the influence of residual gas in PC on NO_x emissions, comparing the emissions results for a scavenged PC and a non-scavenged PC. By means of simulation and experimentation, was found that the residual gas in PC does not affect the exhaust NO_x at any given air-fuel ratio. These results were explained by the fact of NO_x concentration in PC is little reflected in exhaust gas, due to the small amount of burned gases in PC.

Jarosinski et al. [11] studied a homogeneous prechamber engine with a catalytic insert in a Diesel engine modified to the Otto cycle. The catalytic insert is kept at temperatures high enough to secure catalytic reactions and ignition of the charge in MC. Results shows that in addition to the extend operation range up to λ = 1.65, the application of the catalytic PC reduced the HC emissions up to 50% for partial loads.

Roethlisberger and Favrat [117] investigated the impact of PC volume and nozzle orifices size, number, distribution and orientation on NO_x, HC and CO emissions. They found that a reduction of PC internal volume yields higher CO and HC emissions at constant NO_x emissions. Varying the nozzle quantity, they found that the transition from 6 to 4 orifices resulted in a reduction of approximately 5% and 9% of the CO and HC emissions, respectively, without increases in NO_x emissions. About the orientation of nozzle orifices, when oriented towards the squish region provided a simultaneous reduction of CO and HC emissions.

About the varying of nozzle size, Roethlisberger and Favrat [117] found the existence of an optimal cross sectional area for nozzle orifices, of 18.85 mm^2, which achieves the lowest emissions. However, they found that the reduction of the nozzle orifice cross sectional area had no significant impact to CO and HC emissions. This occurred due to two conflicting effects: the deeper penetration of the gas jets induced by a smaller nozzle orifice cross sectional area tends to reduce the amount of unburned mixture compressed in the combustion chamber crevices. However, the intensification of combustion process by stronger gas jets increases the cylinder pressure, tending to increase the quantity of unburned mixture flowing into the crevices.

Roubaud et al. [118] compared the use of a biogas with natural gas in a turbocharged 6 cylinder engine with homogeneous PC. The use of the simulated biogas, consisting in adding 40% of CO2 in natural gas, provided a 15% reduction in CO emissions and 8% reduction on HC emissions when compared with the use of natural gas, keeping the same NO_x emissions levels.

Gomes [119] tested a 4-cylinder engine provided with three different settings of homogeneous PC. The first setting had four individual ducts from the top to the nozzle of PC with inclination to direct the flow to the intake and exhaust valves. The second setting had a central duct with 6 mm diameter and four nozzles targeted to the intake and exhaust valves. The third setting had a central duct directly interconnected with the main chamber.

Regarding to emissions, Gomes [119] found a reduction in CO emissions, for all operation range, when compared to the original

engine without PC. The HC emissions showed the same trend behavior for all configurations, but with higher values than the original engine. Contrary to the most studies about PC ignition systems, was found that NO_x emissions for PC engine was greater than that obtained with the original one. The author explained that the high NO_x content occurred due to the high temperature in PC and MC, requiring improvements in his project design to promote the effective cooling of prechamber.

Moreira [38,120] compared a 16 valves 4-cylinder engine, operating with gasoline, with and without PC. He found a reduction up to 91% on CO emissions, justified due to the increased availability of oxygen to join to CO and form CO2. A reduction up to 67% was found on NO_x emissions, justified due to the lower temperature in MC when using PC. On the other hand, HC percentage was usually higher. Author explained that this occurred due to the great squish area, increasing the heat transfer to the chamber walls, being also favored by the nozzle configuration, which had only one central jet, generating some zones where the flame can be quenched.

It was observed that, in accordance with stratified PC engine, in the most studies CO and NO_x emissions decreased significantly and HC emissions increased. This increase can be attributed to greater turbulence generated by homogeneous prechamber, resulting in a consequent increase in heat transfer, which may lead to the quenching of flame in the MC walls.

Further studies are necessary to determine the actual advantages and disadvantages, in relation to emissions, of using prechamber with stratified or homogeneous mixtures, as the last-mentioned becomes attractive for its structural simplicity.

5. Conclusions

Lean or ultra-lean burn is a tendency to improve the efficiency of internal combustion engines. On this, use of prechamber systems shows to be a viable alternative, adding small costs to the project and expressing excellent combustion and emission results, operating in ultra-lean conditions with lambda levels near to 2.5. As the major current concern is to achieve the pollutant limits imposed by legislation, results are especially promising when used in conjunction with other technologies such as EGR and supercharging. Despite of the costs, stratified charge systems points to better results in emissions as compared to homogeneous systems. Researchers pointed to NO_x levels near to zero due to the lower peak of combustion temperature, residence time at high temperatures, and higher oxygen availability. Carbon monoxide reductions have also been found as a consequence of sufficient oxygen available to complete the reactions. HC emissions were higher in most studies about PC systems, being attributed to the increased crevice and surface area caused by inclusion of an additional combustion chamber. Then, use of prechamber in SI engines is a promising alternative to reduce fuel consumption and consequently exhaust emissions, however, it is necessary to improve technologies able to neutralize the negative impacts generated by addition of prechamber, as the increase in erosion of engine components and in HC emissions. Computer modelling and simulations has been proven to be a great tool for prechamber systems development. It can provide high quality results with less investment and, in most cases, in a shorter time, when comparing to experimental tests. Yet, there are not much numerical studies for this specific field, thus indicating a great area that still need to be explored in order to advance in lean combustion with prechamber systems.

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